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Pressure Balanced, Low Hysteresis, Finger Seal Test Results

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Summary

The finger seal is a revolutionary new technology in air to air sealing for secondary flow control and gas path sealing in gas turbine engines. Though the seal has been developed for gas turbines, it can be easily used in any machinery where a high pressure air cavity has to be sealed from a low pressure air cavity, for both static and rotating applications. This seal has demonstrated air leakage considerably less than a conventional labyrinth seal and costs considerably less than a brush seal.

A low hysteresis finger seal design was successfully developed and tested in a seal rig at NASA Glenn Research Center. A total of thirteen configurations were tested to achieve the low hysteresis design. The best design is a pressure balanced finger seal with higher stiffness fingers.

The low hysteresis seal design has undergone extensive rig testing to assess its hysteresis, leakage performance and life capabilities. The hysteresis, performance and endurance test results are presented. Based on this extensive testing, it is determined that the finger seal is ready for testing in an engine.

Introduction

The gas turbine industry uses a variety of sealing mechanisms to contain and direct secondary flow into and around components for cooling, and to limit leakage into and from bearing and disc cavities. The function of these seals is very important to the component efficiencies and attendant engine performance.[1]

Most of these seals are labyrinth seals, which are high-leakage seals. In recent years, brush seals have been introduced which have demonstrated significantly reduced leakage, but they are expensive and have exhibited hysteresis and wear difficulties. A new innovative seal design called a "finger seal," recently patented[2] by AlliedSignal Engines (AE), has demonstrated considerably lower leakage than a labyrinth seal and is considerably cheaper than a brush seal. The cost to produce finger seals is estimated to be 40 to 50 percent of the cost to produce brush seals.

The availability of a long life and low-leakage finger seal has many benefits for propulsion gas turbine engines. The most direct benefit would be to replace labyrinth seals with finger seals at locations that have very high pressure drops directly to ambient, typically main engine and thrust balance seals. This can provide a saving of 1 to 2 percent of the engine flow which directly results in 0.7 to 1.4 percent reduction in specific fuel consumption (SFC) and 0.35 to 0.7 percent reduction in direct operating cost (DOC). For

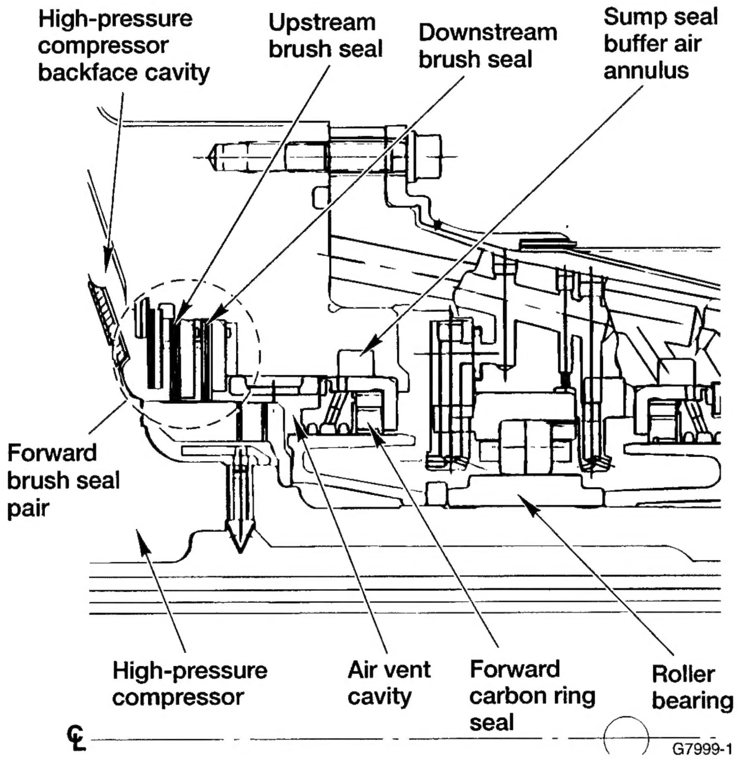


Figure 1.—Typical brush seal arrangement.

an example of their location in an engine, finger seals could be used instead of the brush seals shown in Fig. 1 to seal the high-pressure compressor backface cavity.[3]

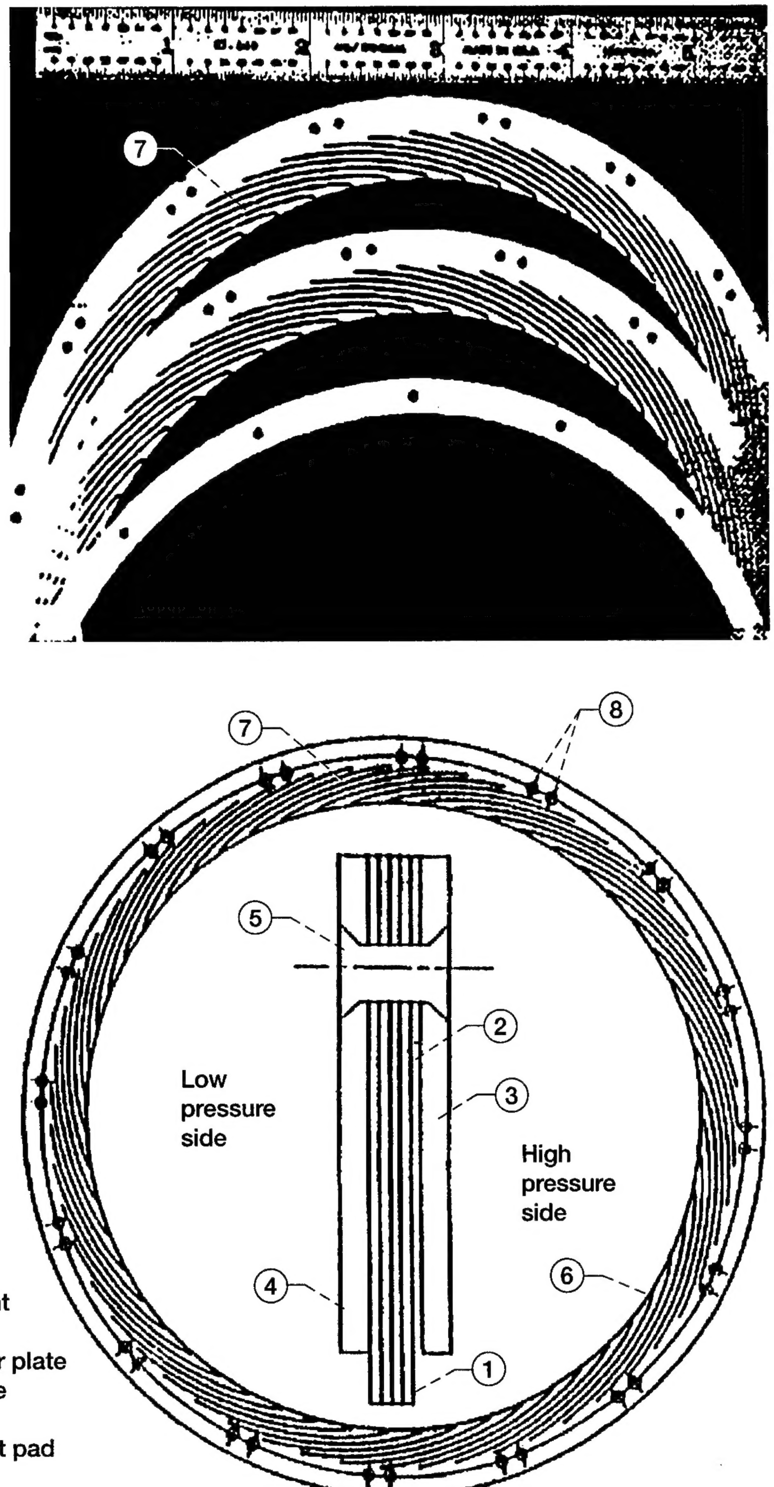
After extensive analytical work and rig testing, a low cost, low hysteresis and low leakage finger seal has been developed. Following a description of the baseline finger seal and rotor and the seal rig test apparatus, a comparison between labyrinth seal and finger seal leakage performance is made and baseline finger seal hysteresis test results are presented. Next, the pressure balanced finger seal is described and its hysteresis, rotor run out, and endurance test results are presented, including leakage and wear performance.

Baseline Finger Seal and Rotor

The finger seal is similar in general configuration to a brush seal, but functions in a very different manner. Instead of a random array of fine wires, the finger seal uses a stack of precision machined sheet stock elements. Each element is machined to create a series of fingers around the inner diameter (Fig. 2). The finger (7) is a slender, curved beam that supports an elongated contact pad (6). Each element also has a series of assembly hole pairs (8). These holes are for the rivets (5) that assemble the seal. The holes are spaced such that when the elements are alternately indexed to the two holes, the spaces between the fingers of one element are covered by the fingers of the adjacent element. Usually a seal is assembled with multiple finger elements (1), a spacer (2), and the forward (3) and aft (4) cover plates. The seal is fitted over the rotating shaft or rotor with a small amount of clearance or interference, depending on the application. Airflow through the seal is impeded by the staggered fingers/pads as well as the radial contact between the rotor and the contact pads. The flexible fingers can bend radially to accommodate shaft excursions and relative growth of the seal and rotor resulting from rotational forces and thermal mismatch.

The finger seal has a low manufacturing cost. The seal laminates are fashioned using the photo-etching process, which is extremely cost-effective. Sheet stock of various alloys and thickness required for the seals is readily available. The photo masking is computer-generated, and location tolerances are accurate to 0.0005 in. The etching process is quite rapid. The riveted assembly does not require any elaborate tooling or assembly process.

The test rotor (Fig. 3) is made of INCO 718 alloy and has a hard coating on the sealing surface. The test rotor was balanced at low speed prior to installation in the rig. The seal and the rotor had an initial radial interference of 0.0072 in. at room temperature.



1. Finger element
2. Spacer
3. Forward cover plate
4. Aft cover plate
5. Rivet
6. Finger contact pad
7. Finger
8. Indexing and rivet holes

Figure 2.—Baseline finger seal and its nomenclature.

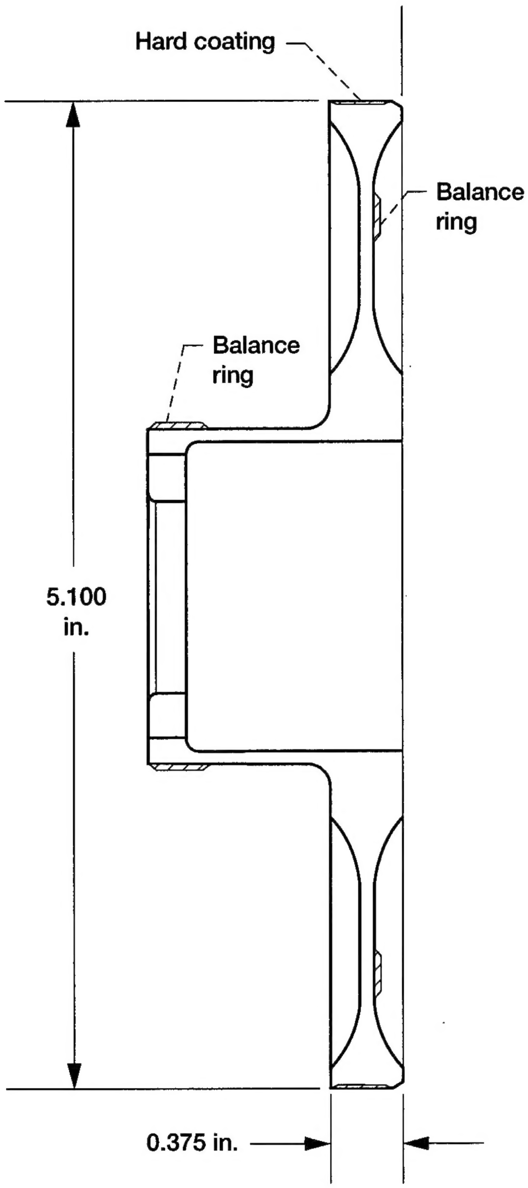


Figure 3.—Finger seal test rotor.

Seal Rig Description

The NASA Glenn seal rig (Fig. 4) was used for testing the finger seal. The rig is capable of operating at shaft speeds up to 45,000 rpm, and 1100, 800 and 70 °F inlet air temperature at 60, 120 and 145 psid air to air pressure differential, respectively. More details about the seal rig can be found in reference 4.

Labyrinth versus Finger Seal Comparison

Figure 5 shows air leakage versus pressure ratio for the finger seal and a four-knife labyrinth seal with various radial clearances. The air leakage is shown as a flow factor, φ, which is defined as:

$$\varphi = \dot{m}\sqrt{\left(T_{avg} + 459.67\right)}\Big/P_H \times D$$

where

$\dot{m}$ = Air leakage flow rate, lbm/sec
T_{avg} = Average air temperature upstream of the seal, °F
P_H = Air pressure upstream of the seal, psia
D = Outside diameter of the seal rotor, in.

Pressure ratio is defined as upstream seal pressure divided by downstream seal pressure.

The finger seal leakage flow factor, from test data, is bounded by a lower and an upper line indicating that leakage will vary between the two limits depending upon the run out of the rotor and the pressure ratio. The labyrinth seal leakage flow factor is an estimate from a well calibrated computer code that is proprietary to AE.

Figure 5 shows the finger seal has 20 to 70 percent less leakage than a typical four-knife labyrinth seal with a 0.005 in. radial clearance, which is a typical clearance for a labyrinth seal near an oil sump.

Baseline Finger Seal Hysteresis Test

The baseline finger seal was tested for hysteresis in leakage flow factor due to changes in speed. This test was run at a constant pressure and temperature, while speed was ramped up from 0 to 45,000 rpm and then down to 0 rpm in 5 to 10 krpm increments. The air leakage flow was measured as a function of speed. The speed ramp up and down cycle was repeated three times. Figures 6(a), (b), and (c) show the measured flow factor, φ, versus rotor speed for cycles 1, 2, and 3, respectively. The plots show that the flow factor was considerably lower during the speed ramp up as compared to the speed ramp down for the first two cycles (Figs. 6(a) and (b)). However, in the third cycle the leakage flow factor had little variation between the increasing and decreasing speed ramps (Fig. 6(c)). Compliant by design, finger seals accommodate centrifugal growth of the rotor, rotor runout, seal offset, and thermal mismatch. During speed ramp up, the fingers move out radially due to the factors mentioned above. Apparently on speed ramp down the fingers did not recover to their original position, which is evidenced by the higher flow factors. It takes a couple of cycles for the fingers to completely move out. Once they do, there is little difference in the flow factors for the increasing and decreasing speed

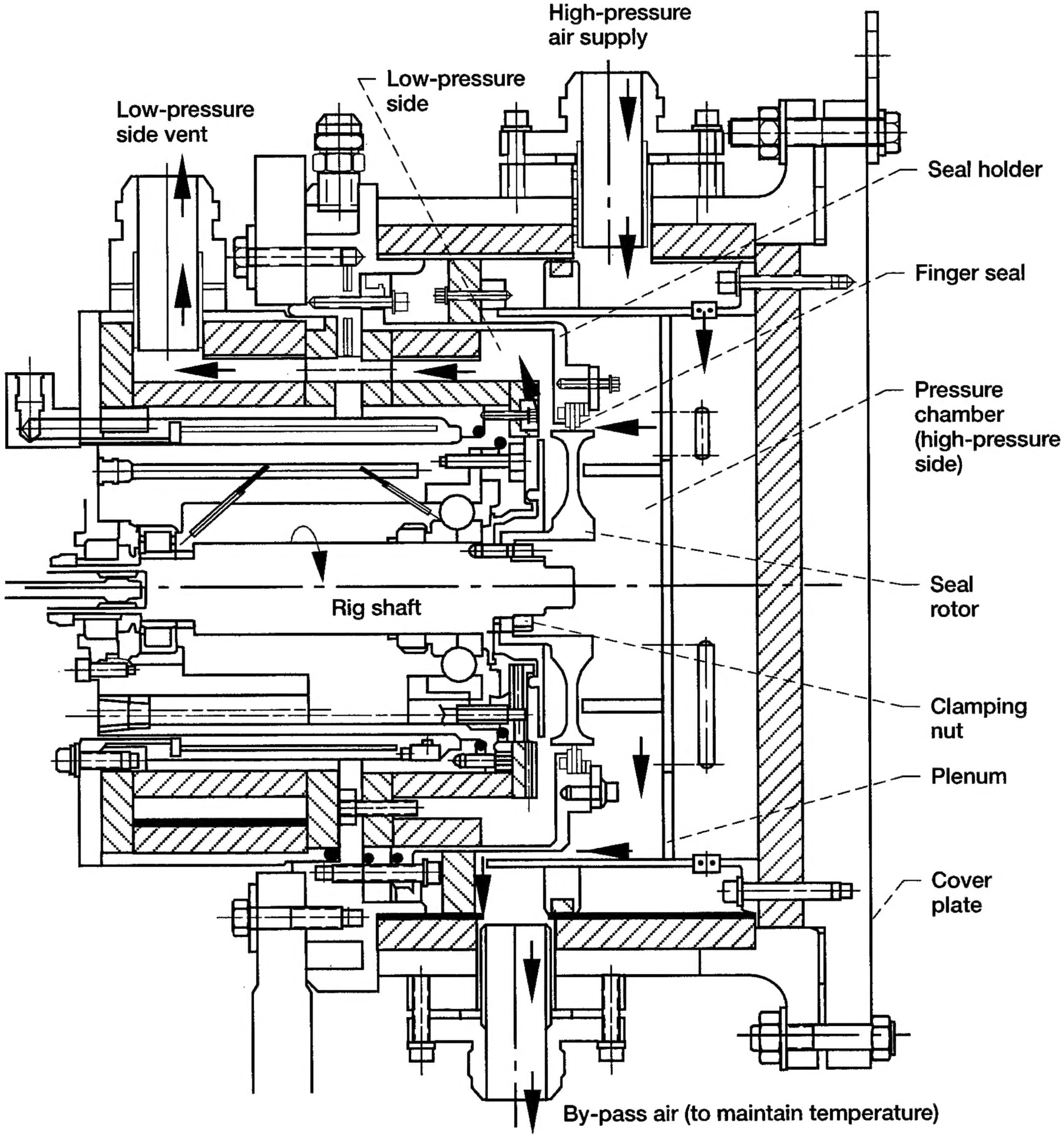


Figure 4.—NASA Glenn seal rig cross section.

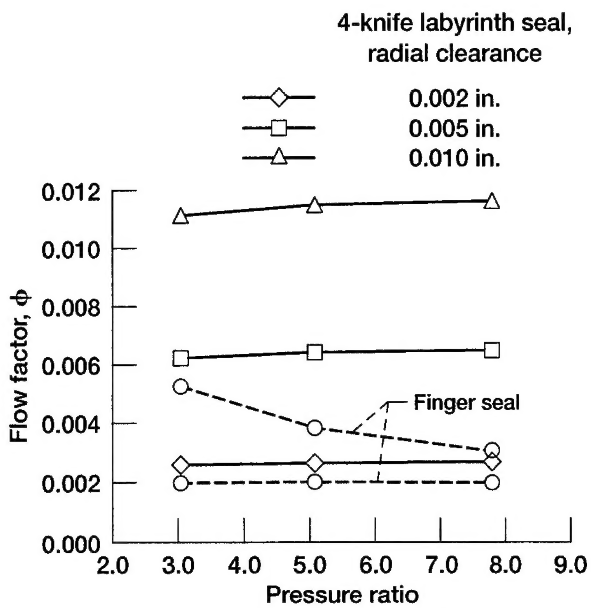


Figure 5.—Comparison of finger and labyrinth seal air leakage performance. Air temperature, 800 °F; shaft speed, 40 000 rpm; seal inner diameter, 5 inches.

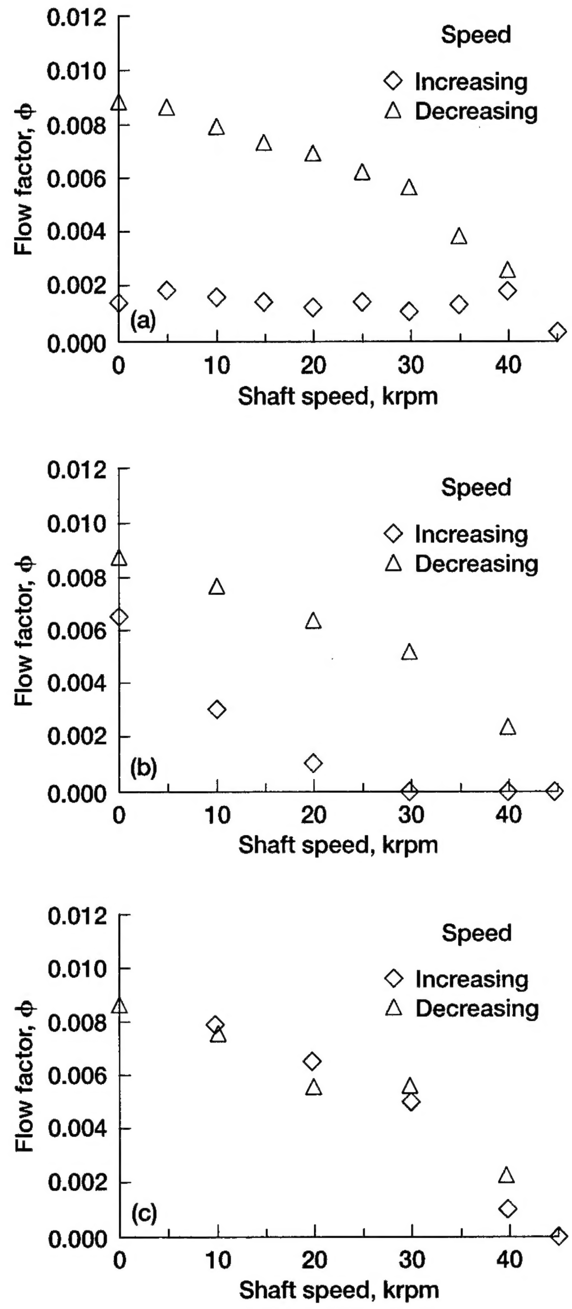


Figure 6.—Baseline finger seal showing significant hysteresis in speed ramp test. Air temperature, 800 °F; pressure drop across seal, 60 psid; (a) speed ramp cycle 1; (b) speed ramp cycle 2; (c) speed ramp cycle 3.

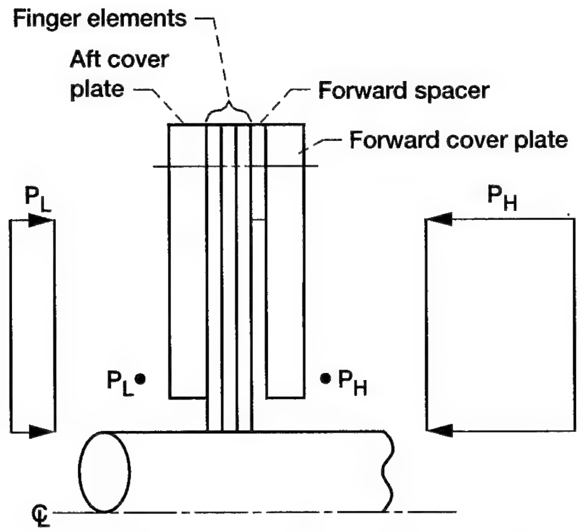


Figure 7.—Baseline finger seal force balance showing areas over which high pressure, P_H, and low pressure, P_L, act.

ramps. The fingers could not recover because the frictional force between the aft cover plate and the fingers was greater than the restoring force in the fingers.

Figure 7 shows the force balance diagram for the baseline finger seal when air pressure is applied. It is evident from the force diagram that a significant force acts on the finger elements. This force is reacted by the aft cover plate. This force causes friction between the fingers and the aft cover plate. The magnitude of this frictional force is much greater than the restoring force available from the fingers, hence once the fingers move out, they can not recover to their original position. The restoring force available from the fingers is a function of the finger laminate thickness, finger length, and material properties.

Hysteresis is a major limitation of the baseline finger seal design, as it leads to inconsistent and, at times, higher air leakage rates.

Pressure Balanced Finger Seal and Rotor

A new finger seal design called "Pressure Balanced Finger seal" (patent pending) (Fig. 8) was shown to be effective in correcting the hysteresis deficiency in the baseline finger seal design. It is similar to the baseline finger seal, except for an additional spacer between the finger laminates and the aft coverplate. This spacer creates a space between the laminates and the aft coverplate and forms a narrow sealing dam near the seal inner diameter. The space created between the laminates and the aft coverplate is connected to the high pressure side of the seal by a series of radial and axial holes and is at an intermediate pressure (P_I) that closely tracks the upstream pressure.

Figure 8 shows the force balance diagram for the pressure balanced finger seal. The net force acting on the seal is a function of high pressure minus low pressure and the areas on which the high and intermediate pressures are acting. The net force on the fingers can be reduced to a very low value by making proper selection of the area on which the high pressure acts. A low net force on the fingers will reduce the friction between the fingers and aft cover plate to a level that prevents binding of the fingers against the aft cover plate. This in turn will allow the finger

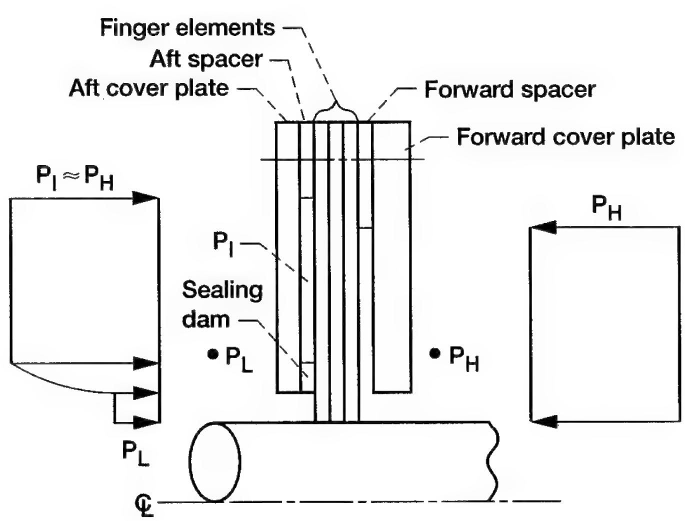


Figure 8.—Pressure balanced finger seal force balance showing areas over which the high pressure, P_H, low pressure, P_L, and intermediate pressure, P_I, act.

contact pads to maintain their original position at all operating conditions. Therefore, this design is expected to provide a consistent low leakage at all steady state and transient operating conditions.

The seal and rotor used for the hysteresis and rotor run out test had an initial radial interference of 0.0009 in. at room temperature. Unlike the concentric rotor used for the baseline finger seal, the rotor used for the pressure balanced finger seal hysteresis and rotor run out tests had a radial run out of 0.0044 in. (0.0088 total indicated run out, TIR) of its outer diameter relative to its inner diameter pilot.

Pressure Balanced Finger Seal Hysteresis Test

The pressure balanced finger seal was tested for hysteresis in the same manner as the baseline seal hysteresis test. The air leakage flow was measured as a function of speed. The speed ramp up and down cycle was repeated three times. Figures 9(a), (b), and (c) show air leakage flow factor (φ) against rotor speed for cycles 1, 2, and 3, respectively. The plot shows that the flow factor was similar for the speed ramp up and speed ramp down for all three cycles. This indicates that the hysteresis has been considerably reduced in the pressure balance design.

Pressure Balanced Finger Seal Rotor Run Out Test

To simulate transient conditions in a flight engine, like traversing critical speeds and fast acceleration and deceleration, a cyclic test was conducted in two segments of 100 cycles each. Each segment involved 5 hours of testing. After the first 100 cycles, the seal and rotor were removed from the rig and inspected. The seal and rotor were then installed back in the rig and tested for another 100 cycles. The seal and rotor were again removed from the rig and given a final inspection.

The operating conditions for the cycle in the first test segment were 2 min at 35,000 rpm, 30 psid and 500 °F followed by 1 min at 10,000 rpm, 30 psid and 500 °F.

The operating conditions for the cycle in the second test segment were 2 min at 35,000 rpm, 60 psid and 800 °F followed by 1 min at 10,000 rpm, 30 psid and 800 °F. The maximum speed achievable with a 0.0044 in. radial run out on the rotor was 35,000 rpm due to rig vibration limits. These speed and pressure conditions were selected to cover the range of expected engine conditions.

Test Results

Figures 10 and 11 show plots of air leakage flow factor (φ) as a function of the number of cycles. For the first 100 cycles (Fig. 10), the leakage flow factor varied from a minimum of 0.0025 to a maximum of 0.0042. In the next 100 cycles of running (Fig. 11), the leakage flow factor varied from a minimum of 0.0027 to a maximum of 0.0046. The lower flow factor corresponds to higher speed and the higher flow factor corresponds to lower speed as expected. At higher speed, due to rotor centrifugal

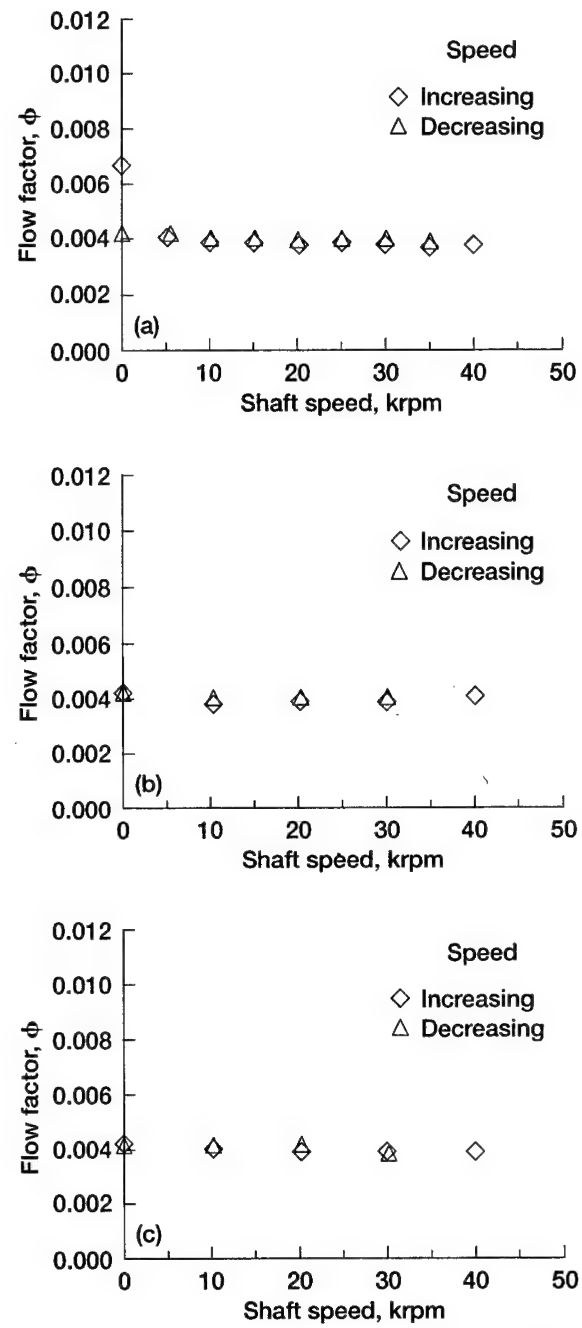


Figure 9.—Pressure balanced finger seal showing very low hysteresis in speed ramp test. Air temperature, 800 °F; pressure drop across seal, 60 psid; (a) speed ramp cycle 1; (b) speed ramp cycle 2; (c) speed ramp cycle 3.

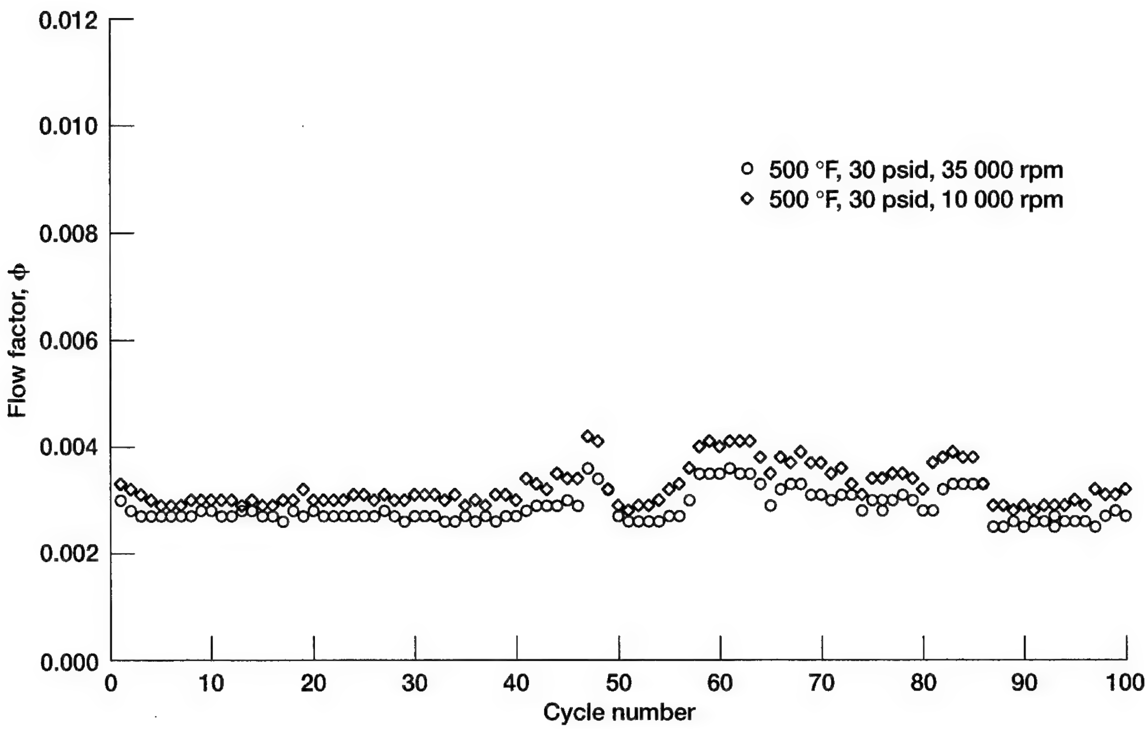


Figure 10.—Segment 1 of pressure balanced finger seal rotor run-out test. Inlet air temperature, 500 °F; pressure drop across seal, 30 psid; speed cycled between 35 000 rpm for 2 minutes and 10 000 rpm for 1 minute.

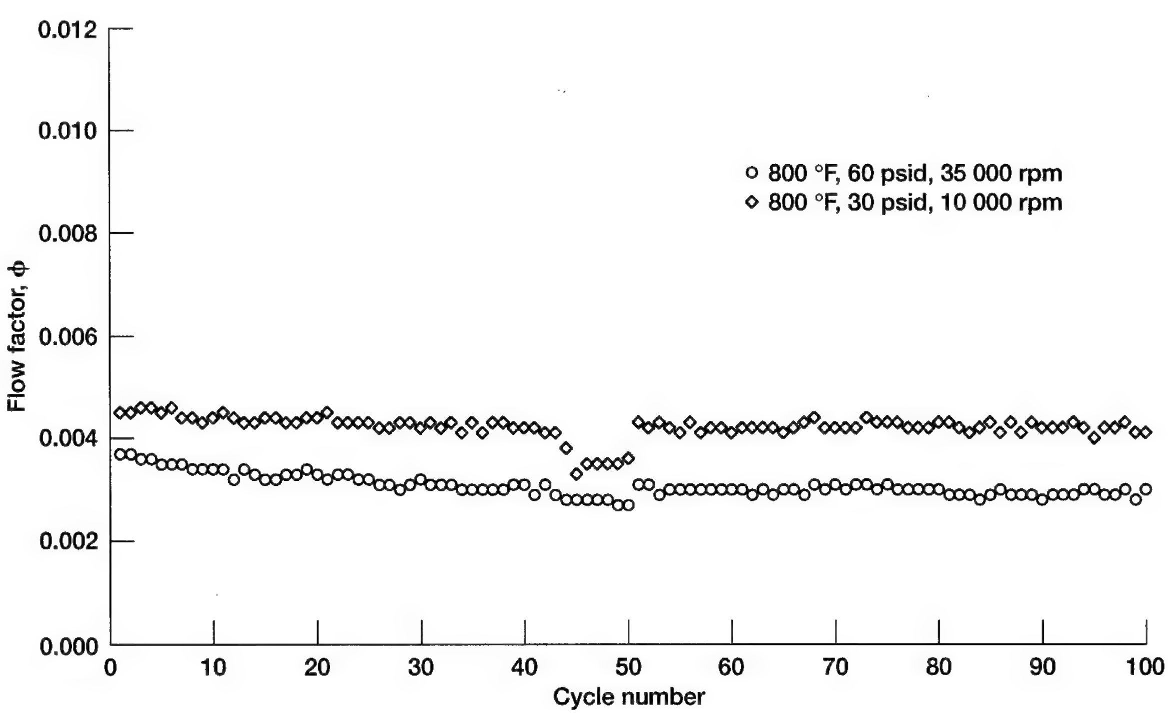


Figure 11.—Segment 2 of pressure balanced finger seal rotor run-out test. Inlet air temperature, 800 °F; pressure drop across seal and speed cycled between 60 psid and 35 000 rpm for 2 minutes and 30 psid and 10 000 rpm for 1 minute.

growth, the seal maintains an interference fit with the rotor, hence the lower flow. Whereas, at lower speed the seal maintains a slight clearance with the rotor, hence a higher flow. For the last fifty cycles, the flow factor generally is lower and stable. It can be concluded that after 150 cycles of running, the flow factor has stabilized.

Inspection Results

The total run time accumulated during rotor run out testing was 10 hr.

After the first segment of tests (5 hr of run time), the finger seal inner diameter (i.d.) was larger than the pretest i.d. by 0.0089 in. After 10 hr of run time, the finger seal i.d. was larger than the pretest i.d. by 0.0131 in. The average finger pad wear after 5 and 10 hr of run time was 0.0053 and 0.0102 in., respectively. The finger pad wear was measured by measuring the pad thickness of the same three fingers (~120° apart) on both the upstream and downstream sides of the seal before and after testing using a microscope. Average wear was computed for the upstream and down-stream sides. The average finger pad wear reported here was the greater of the two averages, which occurred on the upstream side of the seal.

The rotor had a visible wear track on its outer diameter at the seal location.

The maximum depth of the wear track after 5 hours of run time was 0.0061 in. and after 10 hours was the same.

It is anticipated that after a large number of cycles the rate of finger pad and rotor coating wear will become negligible and the finger and rotor interface will reach a line-to-line (no interference, no clearance) contact condition.

Pressure Balanced Finger Seal Endurance Test

A new set of hardware was used for the pressure balanced finger seal endurance test. The rotor had a small runout of 0.0017 in. TIR and the rotor and seal had an initial radial interference of 0.00045 in. at room temperature.

A 120 hr endurance test was performed to evaluate the long term air leakage characteristics of the finger seal. This test was a cyclic test conducted in two segments of 360 cycles each. After the first 360 cycles (60 hr), the seal and rotor were removed from the rig and inspected. The seal and rotor were then reinstalled in the rig and tested for another 360 cycles (60 hr). The seal and rotor were again removed from the rig and given a final inspection.

The operating conditions for the cycle in the first test segment were five minutes at speeds in the range of 42,500 to 35,000 rpm (surface velocity of 945 to 778 fps), 80 psid and 800 °F followed by 5 min at 10,000 rpm, 30 psid and 800 °F. Difficulty with high vibrations in the test rig pre-vented taking the high-speed data uniformly at 42,500 rpm. This segment of testing was conducted on seven different days. The rig was shutdown and cooled each day.

The operating conditions for the cycle in the second test segment were 5 min at 35,000 rpm, 60 psid and 1000 °F followed by 5 min at 10,000 rpm, 30 psid and 1000 °F. This segment of testing was conducted on 8 different days

Test Results

Figures 12 and 13 show a plot of air leakage flow factor as a function of number of cycles. For the first 360 cycles (Fig. 12), the leakage flow factor varied from a minimum of 0.0015 to a maximum of 0.0045. In the next 360 cycles of running (Fig. 13), the leakage flow factor varied from a minimum of 0.0018 to a maximum of 0.006. The lower flow factor corresponds to higher speed and higher flow factor corresponds to lower speed as expected. The flow factor seems to be trending lower and stabilizing between 0.002 and 0.0038 towards the end of second segment of endurance testing. It is also evident that at each day's start of endurance test, flow factors are higher and then gradually settle down to a lower flow factor. This phenomenon may be rig related and may be due to the time required to achieve a steady state rig metal-temperatures. For example, since the only heat source is the incoming air-stream, it takes a period of time for all components including the rotor to reach steady-state temperature. Component temperatures affect their relative sizes thereby influencing the precise rotor-to-seal contact (or interference) condition.

Inspection Results

After 60 hr of run time, the finger seal i.d. was larger than the pretest i.d. by 0.0088 in. After 120 hr of run time, the finger seal i.d. was larger than the pretest i.d. by 0.0096 in. The average finger pad wear after 60 and 120 hr of run time was 0.0057 and 0.0065 in., respectively.

The rotor had a visible wear track on its outer diameter at the seal location. The maximum depth of the wear track after 60 and 120 hr of run time was 0.0030 and 0.0031 in., respectively. It seems that the finger and rotor wear rate decreased with time, indicating that the finger pad and rotor were approaching line-to-line contact at their interface. This is an acceptable level of wear for use in an engine.

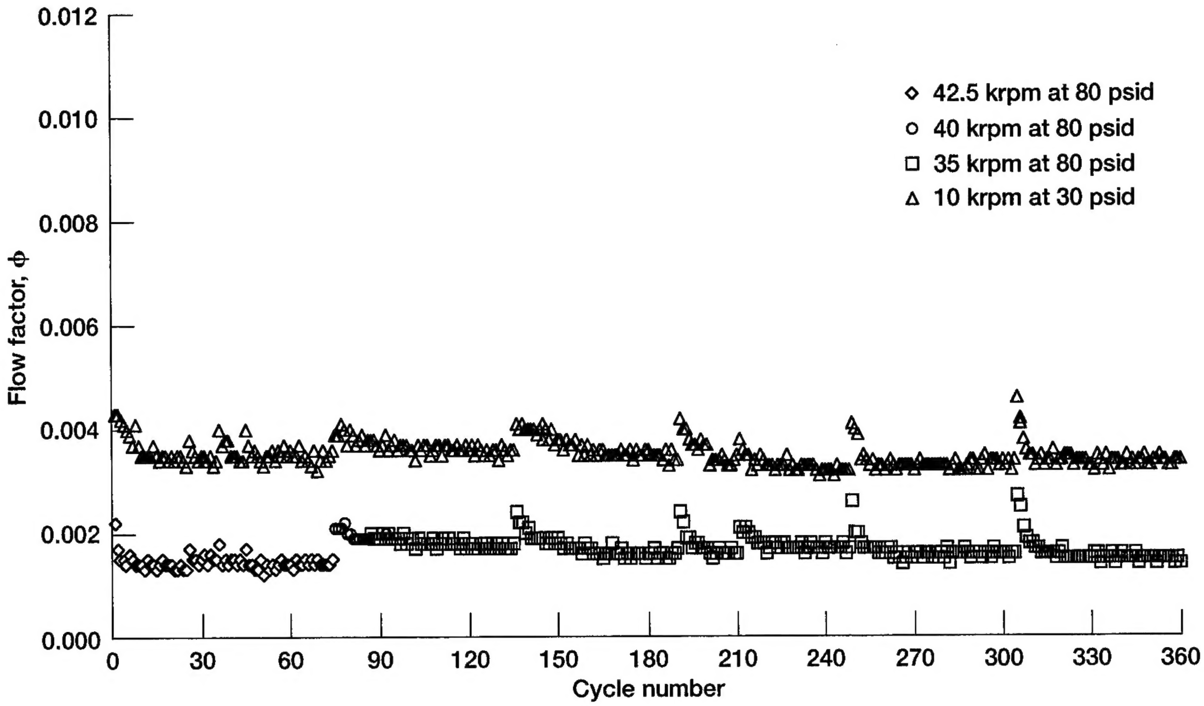


Figure 12.—Segment 1 of pressure balanced finger seal endurance test. Inlet air temperature, 800 °F; pressure drop across seal and speed cycled between 80 psid and 42.5, 40, and 35 krpm for 5 minutes and 30 psid and 10 000 rpm for 5 minutes.

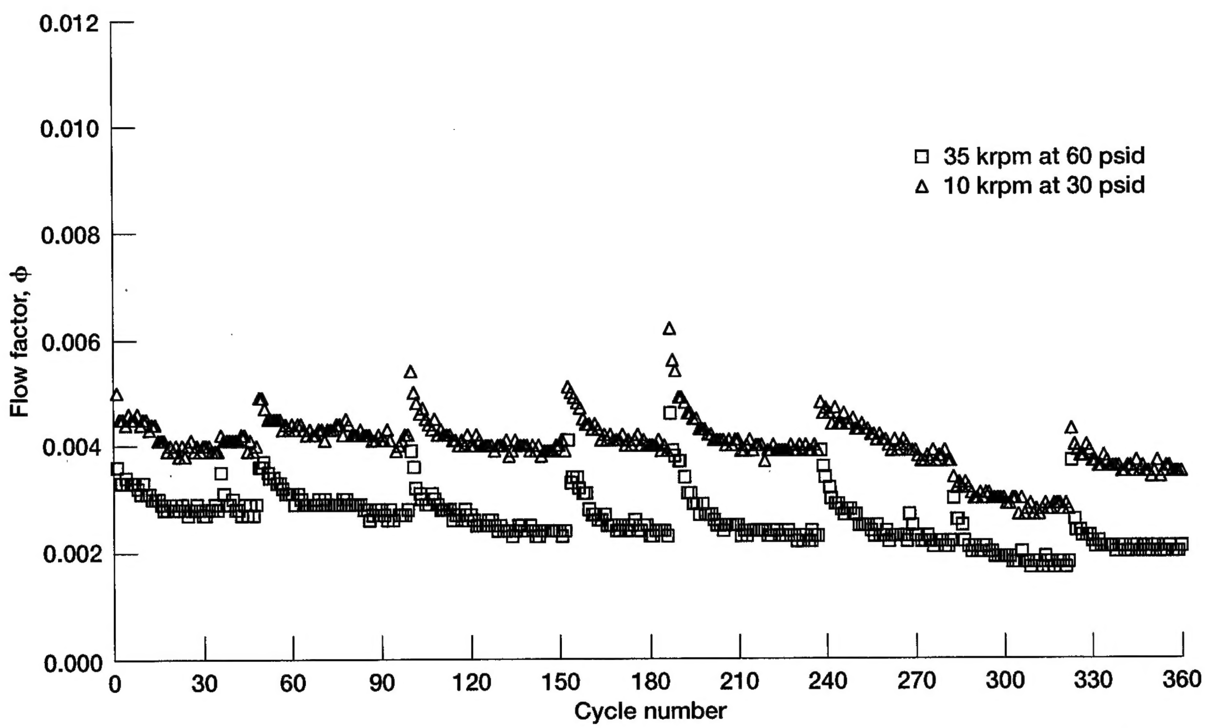


Figure 13.—Segment 2 of pressure balanced finger seal endurance test. Inlet air temperature, 1000 °F; pressure drop across seal and speed cycled between 60 psid and 35 000 rpm for 5 minutes and 30 psid and 10 000 rpm for 5 minutes.

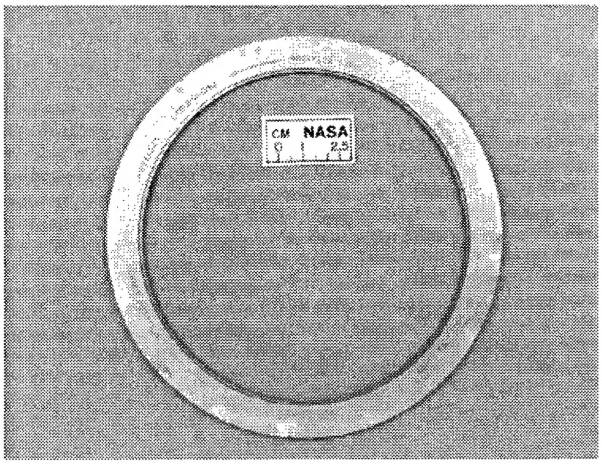


Figure 14.—Overview of pressure balanced finger seal prior to endurance test.

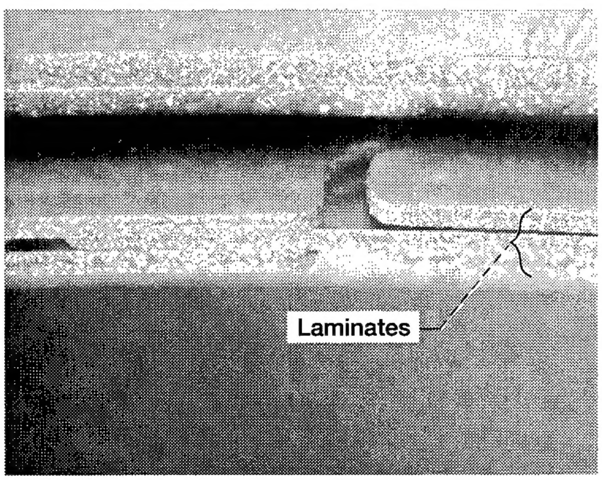


Figure 15.—Magnified view of upstream finger pad i.d. of pressure balanced finger seal prior to endurance test.

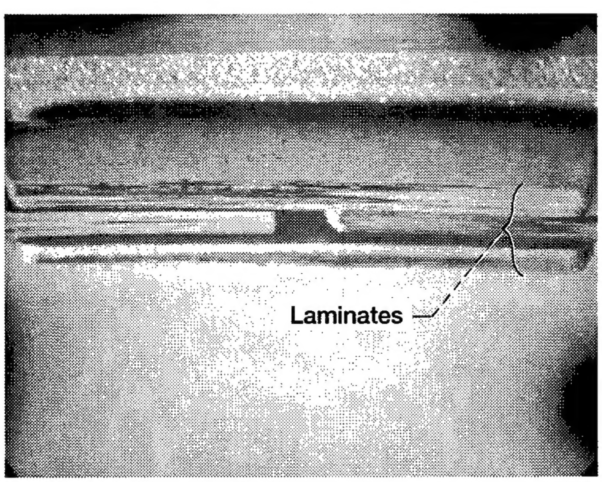


Figure 16.—Magnified view of upstream finger pad i.d. of pressure balanced finger seal after endurance test.

Both the seal and the rotor were in excellent condition after 120 hr of endurance testing. Figures 14 to 16 are photos of the seal before and after the 120 hr endurance test.

Conclusions

1. The seals were manufactured using the low cost fabrication technique of photoetching thin sheet stock. This test program demonstrated that this was a viable approach to making reliable low cost seals.
2. The pressure balanced finger seal design demonstrated very low hysteresis in repeated rig testing.
3. Finger seal air leakage is 20 to 70 percent less than a typical four-knife labyrinth seal with a 0.005 in. radial clearance.
4. Finger seal operation at 778 ft/s, 60 psid and 1000 °F and 945 ft/s, 80 psid and 800 °F was successfully demonstrated.
5. The rotor run out and endurance test results indicate that finger seals have potential for long life applications.
6. Extensive analytical work and rig testing has resulted in a finger seal design that is ready for engine testing.

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13. ABSTRACT ***(Maximum 200 words)***

The finger seal is a revolutionary new technology in air to air sealing for secondary flow control and gas path sealing in gas turbine engines. Though the seal has been developed for gas turbines, it can be easily used in any machinery where a high pressure air cavity has to be sealed from a low pressure air cavity, for both static and rotating applications. This seal has demonstrated air leakage considerably less than a conventional labyrinth seal and costs considerably less than a brush seal. A low hysteresis finger seal design was successfully developed and tested in a seal rig at NASA Glenn Research Center. A total of thirteen configurations were tested to achieve the low hysteresis design. The best design is a pressure balanced finger seal with higher stiffness fingers. The low hysteresis seal design has undergone extensive rig testing to assess its hysteresis, leakage performance and life capabilities. The hysteresis, performance and endurance test results are presented. Based on this extensive testing, it is determined that the finger seal is ready for testing in an engine.

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